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DIESEL ENGINE TURBOCHARGERS: ANALYSIS AND TESTING

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Abstract: *Turbochargers have been and still are a necessity when we talk about internal combustion engines downsizing. The turbocharger is the engine part which produces multiple vibrational and noise phenomena during functioning. Regarding the optimization related to noise and vibration produced by turbochargers, in this paperwork, there were accomplished several software simulations and laboratory tests. The simulations were accomplished in order to highlight the main natural frequencies that could occur during the turbocharger functioning. For validating the data sets obtained by simulations there were performed test rig tests focused on rotordynamic stability of turbochargers.*

Key words: *turbocharger, rotordynamics, natural frequencies, modal analysis.*

1. Introduction

From the various methods used for supercharging internal combustion engines the most used one, involving fabrication costs, which are relatively low in comparison with other methods and also due to high adaptability on various engines, is the turbocharger. Modern solutions of turbochargers use, in order to sustain the turbocharger shaft, hybrid rolling bearings, while classical turbochargers use hydrodynamic bearings. The main advantage presented by using hybrid rolling bearings, consists on a higher durability unlike the case of hydrodynamic bearings. These rolling bearings involve also some disadvantages which are related to higher noise produced during functioning, than the hydrodynamic bearings. Also some tests highlighted the fact that turbochargers rotors sustained by rolling bearings produce higher vibrations than the turbochargers with classical hydrodynamic bearings. The noise and vibrational phenomena produced at the level of turbochargers rotors, which are the only elements in motion, are transmitted further to the housing, where there are amplified, resulting in a complex vibrational phenomena, in some cases the generated vibration could be captured by human ear.

2. Objectives

The main objectives of this paperwork is focused on highlighting some aspects related to turbocharger rotordynamic stability and also on highlighting aspects related to vibrational phenomena that occur at the level of turbocharger entire assembly. For accomplishing the main objectives there were involved software simulation methods and also test rig tests accomplished with specialised vibration

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monitoring equipment. For accomplishing software simulations there was used Matlab-Simulink software solution and also 3D modelling techniques accomplished using Catia V5 software in order to use the generated geometry in simulations (modal analysis). The experimental research was accomplished using two categories of methods: static and dynamic. The static methods are related to accomplishing laboratory modal analysis and dynamic methods refer to validation of some vibrational phenomena on the turbocharger specialised and approved test rig.

3. Simulation of the Dynamic Behaviour of the Turbocharger Rotor

Been the main source of noise and vibration the turbocharger rotor needs a special treatment. In the first step there was accomplished a mathematical model of the turbocharger rotor sustained in hydrodynamic bearings, model which is capable on highlighting the rotational movement amplitudes of the rotor. The mathematical model [2], [4], [10] was designed in such ways that the rotor stability could be controlled by modifying some parameters related to: eccentricity; mass; stiffness and damping [6]. Using Matlab software there were accomplished several simulations for different rotational speeds and bearing stiffness [3], [9]. It could be observed in Figure 1. That at a value of $k = 0$ N/m of bearing stiffness, the

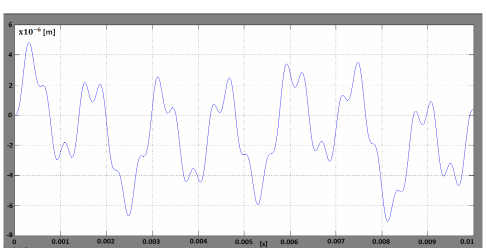

Fig. 1. *The y_2 (displacement of bearing 1 on y direction) - displacement for $k = 0$ N/m*

rotor stability (amplitude) is established on a value which doesn't coincide with the central geometric axis of the rotor.

At a stiffness $k = 1e6$ N/m the rotational movement amplitude can be observed in Figure 2.

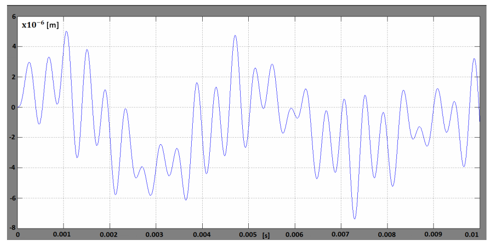

Fig. 2. *The y_3 (displacement of bearing 2 on y direction) - displacement for $k = 1e6$ N/m*

We could observe that also in the case of Figure 1 the rotor is stabilizing its motion on an axis which doesn't coincide with the central geometric axis. In order to highlight the influence that the stiffness has over the motion of the rotor the obtained simulation data is summarized in Table 1.

Displacement for bearings Table 1

Displacement for bearing 1 Y1[m]	Displacement for bearing 2 Y2 [m]	Stiffness value k [N/m]
11.8e-6	11.4e-6	$k = 0$
11.7e-6	11.1e-6	$k = 1$
11.9e-6	11.8e-6	$k = 1e1$
11.95e-6	11.6e-6	$k = 1e2$
12e-6	11.65e-6	$k = 1e3$
11.97e-6	11.35e-6	$k = 1e4$
11.3e-6	11.5e-6	$k = 1e5$
8.1e-6	12.5e-6	$k = 1e6$

4. Modal Analysis of the Turbocharger

For highlighting the main natural frequencies that could appear during the turbocharger functioning it was accomplished the 3D modeling of the turbocharger rotor involving also the bearings, in order to use the modelled parts for accomplishing software modal analysis [1], [7], [11]. The main natural frequencies obtained for the rotor part are as follows:

Mode shapes and frequencies Table 2

Mode shape	Frequency [Hz]
1	409.3
2	410.46
3	666.44
4	669.89
5	858.38
6	921.37
7	5003.1
8	5022
9	5880.1
10	5912.4

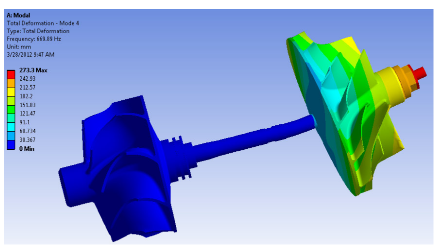

Fig. 3. *Rotor mode shape no.4*

The natural frequencies obtained for the turbocharger bearing, are as follows:

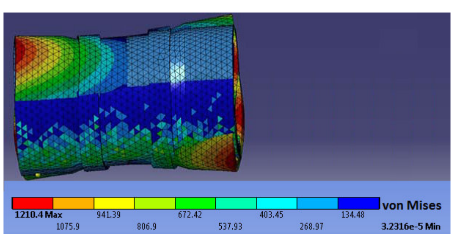

Fig. 4. *Bearing mode shape*

The first mode shape is characterized by the natural frequency of 17820 Hz for the outer ring and 2934 Hz for the entire bearing.

In order to validate the results obtained during the simulations, it was accomplished a modal analysis test using specialized laboratory equipment.

For accomplishing the modal analysis there was used the vibration Pulse 12 platform supplied by Bruel & Kjaer.

The used platform and transducer and impact hammer connection are presented in Figure 5.

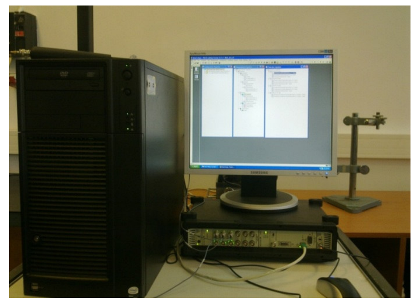

Fig. 5. *Acquisition platform*

The considered chanel connection are as follows:

- for channel 1 the impact hammer;
- for channel 2 accelerometer no. 1;
- for channel 3 accelerometer no. 3.

The accelerometers used are 4507 B types and the impact hammer is 8206-003 type. The method used in modal analysing was the impact hammer method. For a correct test data sets there was performed accelerometer calibration using the 4294 type calibrator supplied by the same producer. The natural frequencies obtained for the turbocharger rotor are highlighted in Figure 6 and the exact values are presented in Table 3.

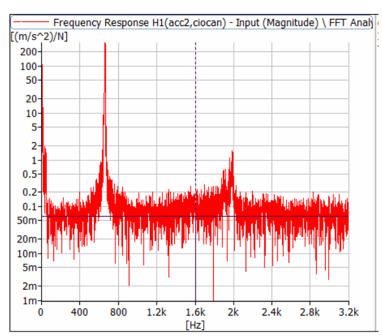

Fig. 6. *Frequency response function for turbocharger rotor*

Acceleration versus frequency Table 3

No	Frequency [Hz]	Acceleration [(m/s^2)/N]	Damping coefficient ζ [%]
1	76	228 m	0.059
2	105	209 m	0.484
3	140	196 m	0.287
4	227	146 m	0.341
5	269	194 m	0.157
6	298	235 m	0.155
7	346	169 m	0.167
8	394	241 m	0.131
9	432	215 m	0.1
10	477	358 m	0.099

The configuration and placement for the transducers and the impact point for the hammer were maintained in the same position as used in the software simulation. The used configuration is presented in Figure 7.

Fig. 7. *Transducer and impact point for the turbocharger rotor*

The configuration and placement for the transducers and the impact point for the hammer were maintained in the same position as used in the software simulation. The used configuration is presented in Figure 8.

The natural frequencies obtained for the turbocharger bearing are presented in Figure 9.

Fig. 8. *Transducer placement and impact point*

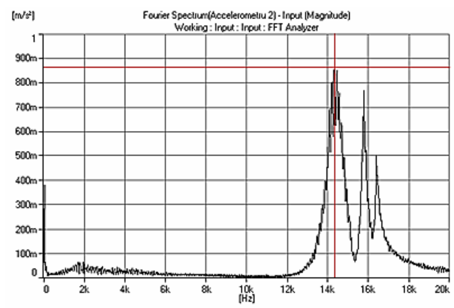

Fig. 9. *Fourier spectrum signal for testes bearing*

For validating the laboratory test performed on the bearing there were used analytical methods. The analytical method used is based on the classical theory for identifying the bearing natural frequencies of the following form [5]:

- Rotating frequency of the inner ring:

$$f_i = n/60; \quad (1)$$

- Rotating frequency of the cage to the outer ring:

$$f_{ce} = n(1 - D_W \cos\alpha / d_m)/120; \quad (2)$$

- Rotating frequency of the cage to the inner ring:

$$f_{ci} = n(1 + D_W \cos\alpha / d_m)/120; \quad (3)$$

• Rotating frequency of the rolling element to the outer ring:

$$f_b = n \cdot d_m \frac{[1-(D_W \cos\alpha / d_m)^2]}{(120 \cdot D_W)}, \qquad (4)$$

where: n - rotational speed; D_w - ball diameter; d_m - medium diameter of the bearing; α - contact angle.

5. Dynamical Testing of the Considered Turbocharger

In order to validate the data sets obtained by simulations it were performed several test rig dynamical tests on a turbocharger specialized and approved test rig. The test rig used in performing the tests was supplied by Schenk. For a correct measurement of the dynamical behaviour of the turbocharger rotor it was considered the simulation of different parameters on the test that are related to: oil temperature; rotor acceleration time, as in real functioning conditions of the tested turbocharger [8]. It is to be mentioned that the tested turbocharger is a new generation of diesel engine turbocharger supplied by Garrett for Mercedes-Benz for a 3.0 litre V6 diesel engine.

In order to establish a reference unbalance, the first test was accomplished by using the turbocharger in undismantled condition,

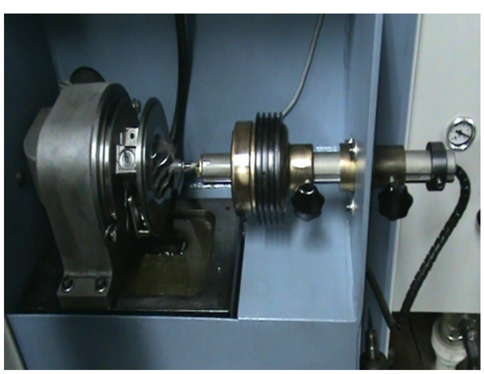

Fig. 10. *Turbocharger mounted on test rig*

maintaining, in this case the same specifications as the factory ones. It were performed several tests in which the speed of the turbocharger rotor achieved a maximum speed value of 86000 rpm. In Figure 10, it could be observed the turbocharger mounted on the test rig.

The rotor precession for 50000 rpm is highlighted in Figure 11.

Fig. 11. *Rotor precession at 50777 rpm*

The rotor precession for 86000 rpm is highlighted in Figure 12.

Fig. 12. *Rotor precession at 86320 rpm*

6. Results and Conclusions

Consulting also the technical specifications supplied for different types of turbochargers we can mention the followings:

• the unbalance at 86000 rpm is 586 um/s, a relatively high value fir this category of turbochargers;

• mathematical models can't highlight the complex functioning behaviour of a turbocharger, due to the complexity of mathematical equations necessary in this cases;

• the vibrations produced by turbochargers are generally generated by rotor precessions;

• vibrational phenomena occurring at the level of the turbocharger are complex and also are strongly influenced by vibrations generated by the functioning of internal combustion engines;

• the rotor stabilizes the movement amplitude over an axis that doesn't concur with the central geometric axis OX of the considered model. The bearings stiffness plays an important role over the rotor precession. The vibrational phenomena of the turbocharger are not only related to the actual vibrations of its component parts, but also to the vibrations that are induced by the functioning of the internal combustion engine, which is the main source of vibrations at the level of vehicles. Unbalanced masses and eccentricity lead to vibration phenomena increasing.

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